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# **A Flexure-based Tool Holder for Sub- $\mu\text{m}$ Positioning of a Single Point Cutting Tool on a Four-axis Lathe**

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## **ABSTRACT**

A tool holder was designed to facilitate the machining of precision meso-scale components with complex three-dimensional shapes with sub- $\mu\text{m}$  accuracy on a four-axis lathe. A four-axis lathe incorporates a rotary table that allows the cutting tool to swivel with respect to the workpiece to enable the machining of complex workpiece forms, and accurately machining complex meso-scale parts often requires that the cutting tool be aligned precisely along the axis of rotation of the rotary table. The tool holder designed in this study has greatly simplified the process of setting the tool in the correct location with sub- $\mu\text{m}$  precision. The tool holder adjusts the tool position using flexures that were designed using finite element analyses. Two flexures adjust the lateral position of the tool to align the center of the nose of the tool with the axis of rotation of the B-axis, and another flexure adjusts the height of the tool. The flexures are driven by manual micrometer adjusters, each of which provides a minimum increment of motion of 20 nm. This tool holder has simplified the process of setting a tool with sub- $\mu\text{m}$  accuracy, and it has significantly reduced the time required to set a tool.

## **KEYWORDS**

Diamond turning, tool setting, tool holder, single point diamond tool, precision machining, meso-scale machining

## **OVERVIEW**

Cutting parts on a lathe is one of the oldest and most versatile machining methods. Modern versions of the lathe include four-axis diamond turning machines, which incorporate a rotary table (B-axis) that allows the cutting tool to swivel with respect to the workpiece to enable the precise machining of complex workpiece forms. Many of these machine tools provide motion accuracy on the order of tens of nanometers, but in order to machine parts accurately on these lathes, the cutting tool must be positioned precisely at the correct height and in the correct lateral position on the rotary table. Setting the tool correctly requires special hardware that allows adjustments of the tool position with the required level of precision over an adequate range of motion in three degrees of freedom. This document describes the design and operation of a tool holder for precisely positioning a single point diamond tool on the B-axis of a four-axis diamond turning machine. The device is designed to be used on a commercial diamond turning machine and can adjust the cutting tool position in three orthogonal directions with 20 nm resolution. The tool holder uses flexure mechanisms to move the cutting tool, so there are no mechanical components that slide relative to each other, and therefore there are no potential detrimental effects from trapped friction. It also has parasitic error motion of less than 0.1  $\mu\text{m}$ , so it enables tools to be set relatively rapidly to machine complex components with sub- $\mu\text{m}$  accuracy. A photograph of the tool holder appears in Figure 1.

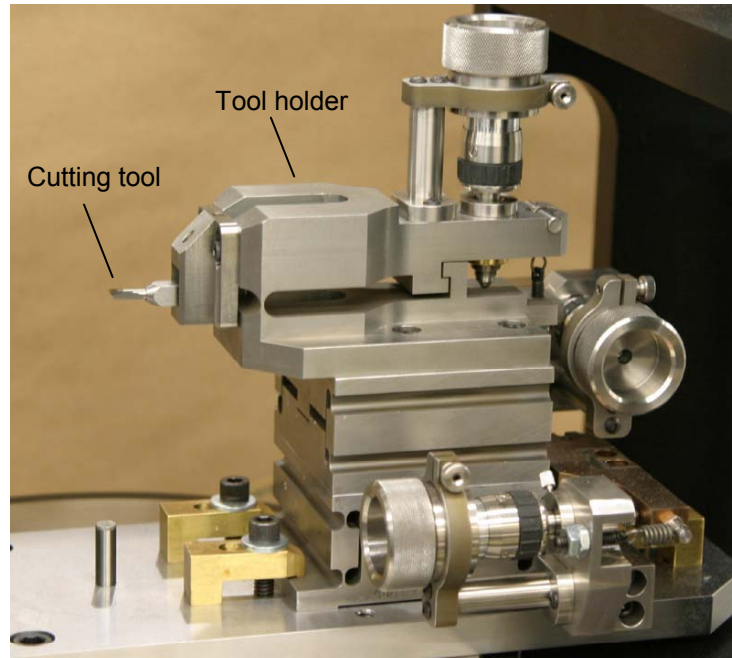


Figure 1. Flexure-based tool holder

## BACKGROUND

The four axes of a precision lathe are illustrated in Figure 2. The spindle of the lathe is mounted on the x-axis, which moves the spindle and the workpiece toward or away from the operator. The z-axis supports the B-axis and the cutting tool, and it moves the cutting tool toward or away from the workpiece in the direction parallel to the spindle axis. The spindle itself is referred to as the C-axis and can be operated in either a free-spinning mode or in an angular positioning mode. The B-axis is a rotary table mounted on top of the z-axis and rotates the tool in the xz-plane so that it can swivel with respect to the workpiece.

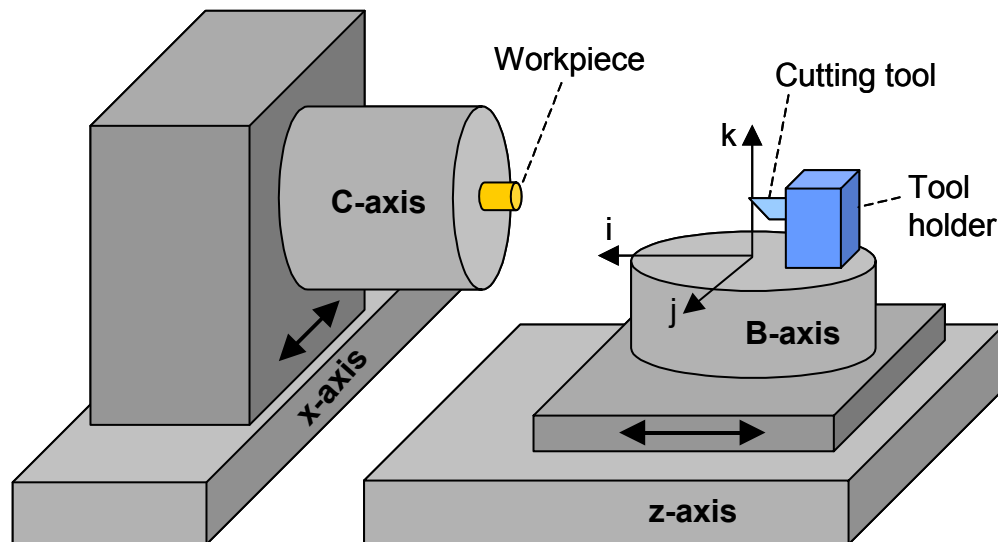


Figure 2. Schematic illustration of a T-based four-axis diamond turning machine

The B-axis of the machine tool is used when machining parts with a complex shape, such as aspheres or oddly shaped lenses, and it is a crucial element to machining parts with both interior and exterior features, such as components for a double shell target [1]. Machining components with sub- $\mu\text{m}$  accuracy on a four-axis lathe is not a trivial task, and there are several approaches that could be used to achieve the required accuracy. This study is concerned with the case in which the nose of the tool has a circular cutting edge with a conical clearance. The approach used in the current study is to position the center of the nose of the single point diamond tool along the axis of rotation of the B-axis. Aligning the tool properly requires a special tool holder that is mounted on the platform of the B-axis. To describe relative motion of the tool with respect to the B-axis platform, a coordinate system is defined that is fixed to the B-axis platform. The i, j, k coordinate system has its origin directly on the axis of rotation of the B-axis. The i and j axes are parallel to the B-axis platform, and the k axis is parallel to the axis of rotation of the B-axis. The k direction is often referred to as the “height” direction. The tool holder maneuvers the tool in the i and j directions to center the tool nose with the axis of rotation of the B-axis, and in the k direction to align the height of the cutting edge with the spindle axis.

In general, the accuracy of a machined workpiece surface is limited by the accuracy with which the tool is set, which often requires that the tool be positioned precisely on the B-axis. However, many commercial machine tools employ special procedures that can tolerate tool setting errors of several  $\mu\text{m}$  or more. Most industrial diamond turned parts machined on four-axis machine tools are either relatively large or have shallow profiles, such as lenses or molds. When machining parts such as these, an interferometer or displacement probe can be used to measure the profile of the machined part to map the error of the surface figure. The machine tool can then compensate for any workpiece profile error to machine the desired profile. This method corrects for any tool setting errors in the i and j directions, so there is no need to set the tool exactly on the axis of rotation of the B-axis. It is only necessary to set the tool at the correct height in the k direction. Commercial tool holders allow adjustments in the i and j directions using sliding plates that are preloaded with set screws and adjusted with differential screws. These mechanisms are not designed to allow sub- $\mu\text{m}$  positioning of the tool in the i and j directions, but they provide a simple means of positioning the tool within a few  $\mu\text{m}$  of the B-axis.

Commercial tool holders exhibit some cross-talk, meaning that every time the tool is moved in one direction, it can move by more than 1  $\mu\text{m}$  in the orthogonal directions, which complicates setting the tool with sub- $\mu\text{m}$  accuracy in the i, j, and k directions simultaneously. In addition, each time the tool is adjusted, friction becomes trapped in the sliding components of the tool holder. If proper measures are not exercised, this trapped friction can be released in the form of incremental slipping of the sliding plates. This slipping can cause the tool to move several  $\mu\text{m}$  over the course of a few days, which further complicates the issue of keeping the tool centered precisely on the B-axis.

The commercial tool holders work well for their intended method of operation, when workpiece error compensation methods can be used. In contrast, error compensation methods are difficult to apply to meso-scale components with complex three-dimensional shapes, such as components for laser targets. These components often contain both interior and exterior features, or they are too complex or too fragile to be reliably measured. Therefore, the commercially available tool holders are not appropriate for certain complex meso-scale components with sub- $\mu\text{m}$  profile requirements.

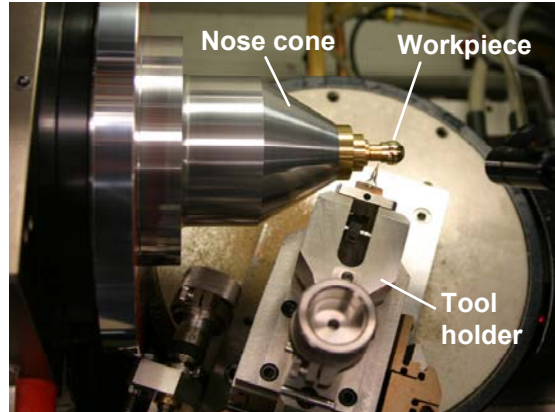
The profiles of meso-scale components with complex three-dimensional shapes must be machined correctly without performing direct, comprehensive metrology of the workpiece. Therefore, obtaining the correct workpiece form requires that the cutting edge of the tool traverse the correct path along the workpiece surface. One way to accomplish this task is to precisely position the nose of the tool on the axis of rotation of the B-axis. However, rather than placing the tool in a precise location along the B-axis, it would also be possible to establish a virtual axis in the k direction that passes through the actual location of the center of the tool nose. The machine tool could be programmed in this virtual, tool-centered coordinate system by using combined motion of the x, z, and B axes to produce the desired motion of the tool relative to the center of the tool nose. Such an approach would require only that the actual location of the tool be known, but it would not require that the tool be positioned in any particular location. The advantage of programming in a virtual coordinate system is that commercially available tool holders could be used to position the tool adequately. However, many commercially available four-axis diamond turning machines do not currently have the capability to allow programming in a virtual, tool-centered coordinate system. Rather than creating a new operator interface and modifying the control system of the diamond turning machine to create a virtual, tool-centered coordinate system, it was more practical and economical in the current study to create a new tool holder with the capability to position the center of the tool nose on the axis of rotation of the B-axis of the diamond turning machine with sub- $\mu\text{m}$  precision.

The approach of employing a tool holder capable of sub- $\mu\text{m}$  tool positioning is not new. Highly precise tool holders have been used for decades to machine meso-scale components with complex three-dimensional shapes on four-axis lathes. For example, several of the precision four-axis machine tools that were designed and constructed at Lawrence Livermore National Laboratory in the 1970's and 1980's use custom tool holders that allow the tool to be oriented along the axis of rotation of the B-axis with nanometer precision. One such tool holder manipulates the tool in the i and j directions using preloaded crossed-roller bearing stages for coarse motion, and piezo actuators for fine motion. The piezo actuators provide only a few  $\mu\text{m}$  of motion but can manipulate the tool with a precision on the order of tens of nanometers. The tool is moved in the k direction by a hydraulic lift that provides nm positioning precision. Unfortunately, this tool holder is too large for modern commercial diamond turning machines. It is also a complicated design that requires an expensive control system to drive the piezo actuators, and it would be expensive to adapt for use on commercial machine tools. Therefore, a new tool holder was designed specifically for implementation on commercial four-axis precision diamond turning machines.

## **DESIGN OF THE TOOL HOLDER**

### **Size Requirements**

The tool holder was designed to fit inside a well defined envelope that is set by the dimensions of commercial four-axis diamond turning machines. The tool holder must align the cutting edge of the tool at the precise height of the axis of the machine tool spindle, and it must align the center of the tool nose with the axis of rotation of the B-axis. The tool holder must be an appropriate size for the B-axis platform, and it must also have sufficient clearance on the side surfaces to avoid contact with the nose cone mounted on the machine tool spindle when the B-axis is rotated through angles of up to  $\pm 110^\circ$ . Figure 3 depicts the clearance between the tool holder and the nose cone when the B-axis is oriented at an angle of  $110^\circ$ .



**Figure 3. Top view of the tool holder oriented at 110° with respect to the spindle**

## **Material**

The tool holder will be used when machining with a variety of both water-based and oil-based cutting fluids, and it must have adequate corrosion resistance for this environment. The tool holder must also have good thermal stability and a relatively small coefficient of thermal expansion. In order to obtain the desired corrosion resistance, high strength, modulus of elasticity, and coefficient of thermal expansion at a reasonable cost, the flexures were made from type 403 martensitic stainless steel. Annealed type 403 stainless steel has a tensile yield strength of approximately 275 MPa, which can be increased significantly by heat-treating [2]. This material has a coefficient of thermal expansion of approximately  $10 \times 10^{-6}/^{\circ}\text{C}$ , and an elastic modulus of 200 GPa, which is typical of most steels.

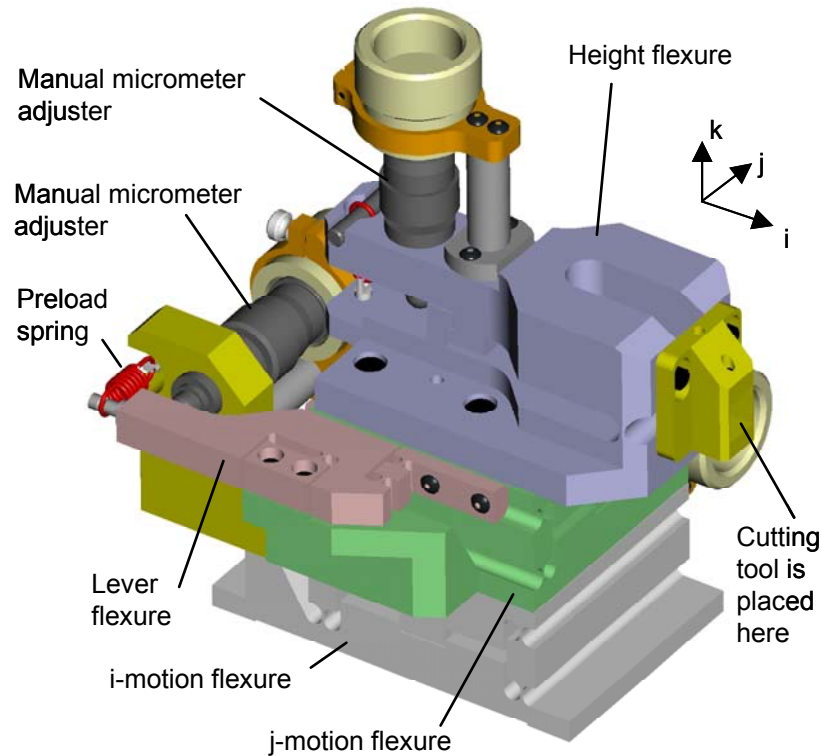
## **Actuators**

To achieve the required level of accuracy in machined workpieces, the tool holder must be able to set the single point diamond tool on the B-axis with sub- $\mu\text{m}$  accuracy. To make the tool holder as inexpensive and as simple as possible to operate, it uses manual adjusters similar to micrometers to provide motion in three orthogonal directions. These high precision adjusters avoid all of the complications associated with hydraulic drivers or electronic actuators. While there are many types of commercially available actuators that can achieve nm level precision, most are significantly more expensive than manual actuators, require expensive control systems, and can only be used by trained operators. Manual high precision adjusters have the advantage of being relatively straightforward to operate, so operators can learn to use them quickly. The actuators selected for the tool holder have a specified minimum increment of motion of 20 nm, a range of  $\pm 150 \mu\text{m}$ , and a specified load capacity of 100 N [3].

## **Flexure Mechanisms**

To avoid the potential problems associated with trapped friction in the hardware, all of the motion in the tool holder is achieved through flexure hinges [4], which rely on elastic deformation of a monolith. Thus, there are no sliding parts and no frictional interactions between components. Therefore, there is no potential for trapped friction to be released from the mechanisms and cause unintended movement of the cutting tool. Flexure hinges are usually more precise and stable than sliding elements or rolling element bearings. The tool holder incorporates three core flexures that allow adjustment of the tool in three orthogonal directions. Each flexure mechanism must allow motion in only the desired degree of freedom, but they must

also be manufacturable at a reasonable cost, have appropriate stiffness, and experience appropriate bending stresses. Each of the flexures on the tool holder was carefully designed using finite element analyses to meet these requirements. A schematic illustration of the tool holder appears in Figure 4.

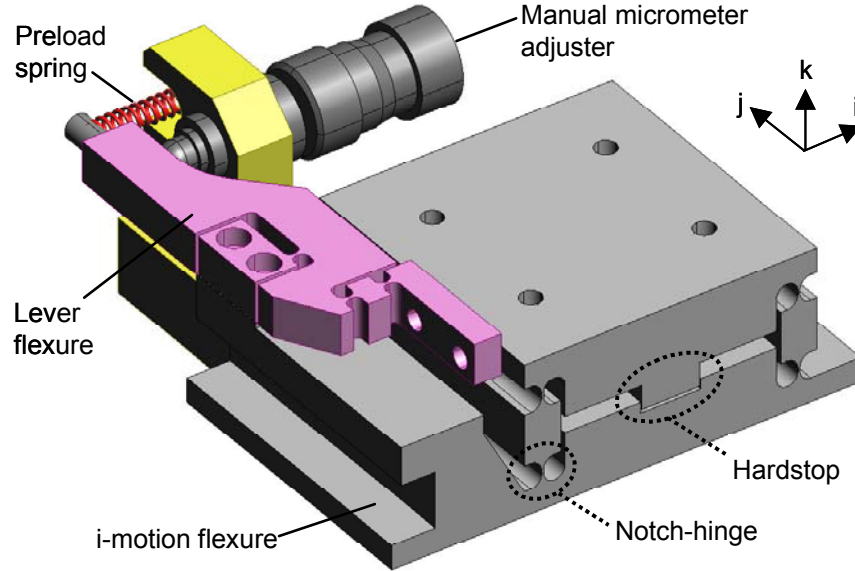


**Figure 4. Schematic illustration of the tool holder**

### **Flexures for Aligning the Tool on the B-axis**

The tool holder includes three stacked flexure mechanisms for manipulating the tool in the  $i$ ,  $j$ , and  $k$  directions. Each of the flexure mechanisms is driven by a manual micrometer adjuster. One adjuster provides motion in the  $i$  direction, and another provides motion in the  $j$  direction. A schematic illustration of the mechanism for adjusting the tool in the  $i$  direction appears in Figure 5. The mechanism for adjusting the tool in the  $j$  direction is nearly identical.





**Figure 5. Mechanism for adjusting the tool position in the *i* direction**

The manual adjuster is connected to a notch-hinge “four bar link” flexure via a lever flexure that reduces the load on the actuator and improves the precision with which the tool can be manipulated. The lever flexure provides a mechanical advantage of  $2.5\times$  and is held against the manual adjuster by a preload extension spring. Therefore, because the manual adjuster has a range of motion of  $\pm 150\ \mu\text{m}$ , the *i*-motion flexure has a range of motion of  $\pm 60\ \mu\text{m}$ . Because of the mechanical advantage, the minimum increment of motion of the *i*-motion flexure is smaller than that of the actuator. The “four bar link” flexure allows motion in only the *i* direction with parasitic error of 0.1% in the *k* direction. Motion is constrained in the other four degrees of freedom.

Each of the four notch-hinges on the flexure has a narrow web that allows the material to bend elastically. Finite element analyses were performed to determine the geometry of the hinge so that it would be manufacturable, have appropriate stiffness, and experience appropriate stresses during operation. The finite element analyses of this flexure hinge were performed using ANSYS with plane strain elements of type PLANE2. The variables considered in the finite element analyses were the width of the flexure in the direction parallel to the hinge, the thickness of the web of the flexure hinge, the notch radius on either side of the web, and the length of the straight section between the fillets. To make the flexure manufacturable at a reasonable cost, the thickness of the narrow web of the flexure hinge was set at 0.5 mm, because this dimension could easily be achieved using a wire EDM process. The design selected after performing numerous finite element analyses uses a notch radius of 2.5 mm and incorporates a straight section of length 1 mm between the fillets, which reduces the bending stiffness of the hinge to decrease the load on the manual adjusters. A finite element analysis indicates that a load of 47.6 N is required to deflect the flexure  $60\ \mu\text{m}$ , which corresponds to full extension or retraction of the manual adjuster. In this condition, the maximum Von Mises stress in the hinge is approximately 73 MPa, which is 27% of the yield strength of the annealed material.

The flexure incorporates a hardstop that prevents it from deflecting too far and plastically deforming the notch-hinges. The motion limit of the hardstop was determined by calculating the displacement required for the Von Mises stress in the hinge to reach 275 MPa, which is the yield strength of annealed type 403 stainless steel. As shown in Figure 6, a finite element analysis

indicates that the maximum Von Mises stress reaches 275 MPa at a displacement of 224  $\mu\text{m}$ . Therefore, the hardstops are designed to limit deflection to 224  $\mu\text{m}$ .

A safety factor is created in the flexures by heat-treating the material at 980  $^{\circ}\text{C}$  and tempering at 650  $^{\circ}\text{C}$ . This heat-treating process should theoretically increase the yield strength of the stainless steel to 586 MPa [2]. Therefore, the stress experienced when the i-motion and j-motion flexures bottom out against the hardstops is 47% of the theoretical yield strength of the heat-treated material. However, it is important to note that the actual yield strength obtained in the central sections of large heat-treated components may be smaller than the theoretical value. For this reason, the minimum possible yield strength, 275 MPa, is used as the design criteria for the flexure, and the material is heat-treated to provide a safety factor.

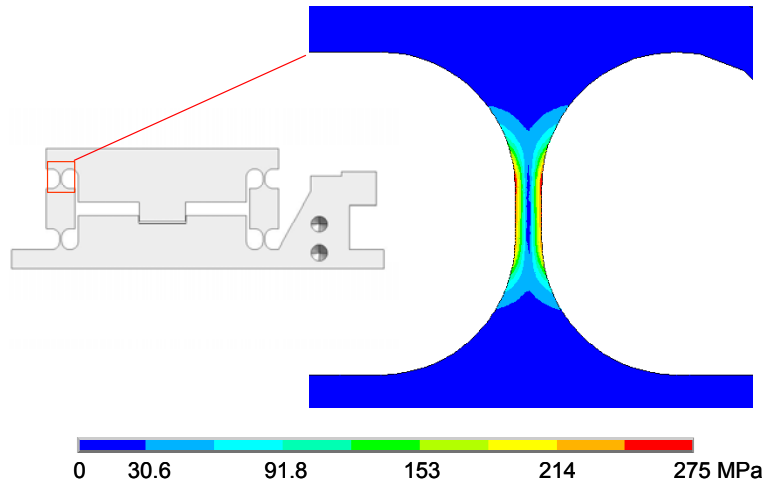
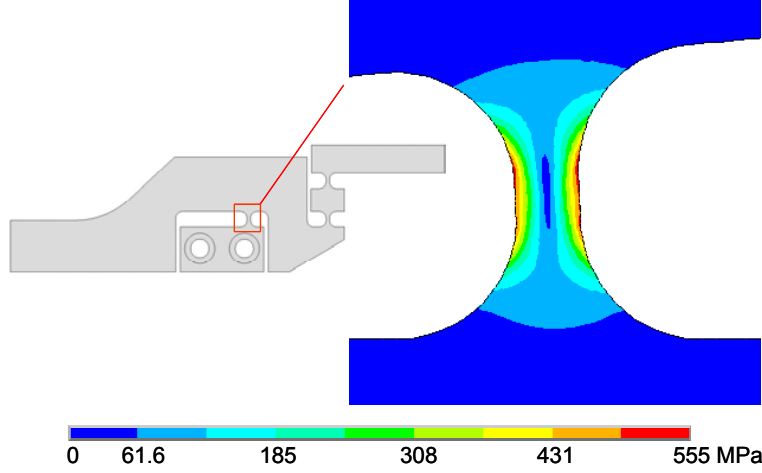


Figure 6. Finite element analysis of the Von Mises stress in the flexure hinge

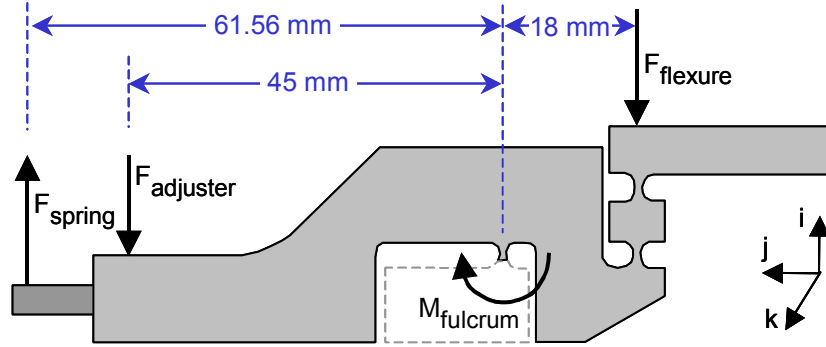
### Lever Flexures

The i-motion and j-motion flexures are connected to the manual adjusters via lever flexures that provide a mechanical advantage of 2.5 $\times$ . These flexures were also designed using finite element analyses. The notch hinge on each lever flexure has a web of thickness 0.8 mm, a notch radius of 2.5 mm, and a straight section of length 1 mm between the fillets. A finite element analysis indicates that when the i-motion flexure deflects 224  $\mu\text{m}$  and bottoms out on the hardstops, the Von Mises stress in the fulcrum notch hinge of the lever flexure is 555 MPa, as shown in Figure 7. To prevent the notch hinge from plastically deforming in this scenario, the lever flexures are heat-treated at 980  $^{\circ}\text{C}$  and tempered at 315  $^{\circ}\text{C}$  to increase the yield strength of the stainless steel to 965 MPa [2]. Note that because of the relatively small size and aspect ratio of these parts, heat treating them to the desired yield strength is not an issue. Therefore, when the i-motion flexure bottoms out on its hardstops, the Von Mises stress in the fulcrum notch hinge of the lever flexure is only 58% of the yield strength of the material.



**Figure 7. Finite element analysis of the Von Mises stress in the fulcrum notch hinge of the lever flexure**

A static analysis of the lever flexure must be performed to determine the preloading force required by the extension spring to keep the lever in contact with the manual adjuster. A finite element analysis indicates that when the i-motion flexure deflects to its maximum working displacement of 60  $\mu\text{m}$ , the fulcrum hinge of the lever flexure experiences an internal bending moment of 0.297  $\text{N}\cdot\text{m}$ . A free body diagram of the lever flexure in this configuration appears in Figure 8.



**Figure 8. Free body diagram of the lever flexure**

Recall that the load capacity of the manual adjusters is 100 N (22  $\text{lb}_f$ ). To avoid damaging the adjusters, the service load on the adjusters is kept below 71.2 N (16  $\text{lb}_f$ ). When the manual adjuster is extended to move the i-motion flexure 60  $\mu\text{m}$  in the i direction, the compressive force between the i-motion flexure and the lever is 47.6 N. A moment balance about the fulcrum hinge of the lever flexure appears in Equation (1).

$$\sum M_{\text{hinge}} = 0 = -(F_{\text{spring}})(0.06156\text{m}) + (F_{\text{adjuster}})(0.045\text{m}) - (M_{\text{fulcrum}}) - (F_{\text{flexure}})(0.018\text{m}) \quad (1)$$

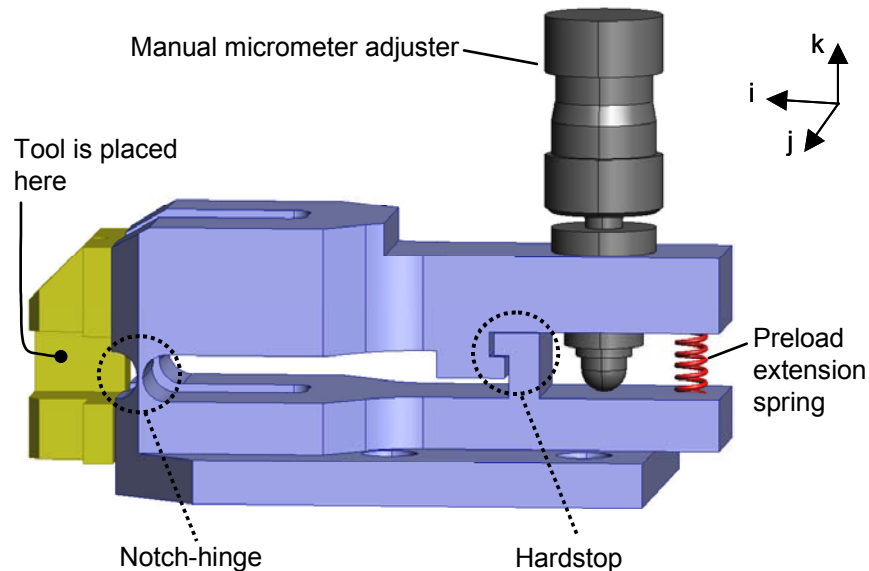
$F_{\text{adjuster}} = 71.2 \text{ N}$ ,  $M_{\text{fulcrum}} = 0.297 \text{ N}\cdot\text{m}$ , and  $F_{\text{flexure}} = 47.6 \text{ N}$ . Therefore, Equation (1) indicates that the required preload force of the spring,  $F_{\text{spring}}$ , is 33.3 N.

When the manual adjuster is retracted to move the i-motion flexure 60  $\mu\text{m}$  in the -i direction, the tensile force between the i-motion flexure and the lever is 47.6 N. For this scenario,  $F_{\text{spring}} = 33.3 \text{ N}$ ,  $M_{\text{fulcrum}} = -0.297 \text{ N}\cdot\text{m}$ , and  $F_{\text{flexure}} = -47.6 \text{ N}$ . Therefore, the load on the adjuster,  $F_{\text{adjuster}}$ , is 19.9 N in this scenario, so there is no risk of damaging the adjuster.

Finite element analyses were performed to determine the stress in the fulcrum notch hinge of the lever flexure when the manual adjuster is fully retracted, and when it is fully extended. The maximum Von Mises stress in these conditions is 159 MPa. Therefore, the stresses that occur in the lever flexure during normal operation are much lower than those that occur when the i-motion and j-motion flexures bottom out against the hardstops. The latter is the limiting case for the design of the lever flexure. During normal operation of the mechanisms that move the tool in the i and j directions, the maximum Von Mises stress in each lever flexure is only 58% of the yield strength of the annealed material, and only 16% of the theoretical yield strength of the heat-treated material.

## Height Flexure

The height of the cutting tool is adjusted to align the cutting edge of the tool with the axis of rotation of the machine tool spindle using a third manual micrometer adjuster that drives another notch-hinge flexure. A schematic illustration of this mechanism appears in Figure 9. The height flexure is actually a lever with a mechanical advantage that decreases the load on the adjuster and improves the precision of the height adjustments. The cutting tool is held in a special clamp at the end of the height flexure. As the tool height is adjusted, the cutting edge traverses a wide arc, moving primarily in the k direction with a relatively small displacement in the i direction. This parasitic error motion in the i direction is a function of the magnitude of the height adjustment and ranges from 0 to 0.1%. The flexure hinge constrains motion in the other four degrees of freedom.



**Figure 9. Mechanism for adjusting the tool in the k direction**

Finite element analyses were performed to design the hinge so it would have appropriate stiffness and experience appropriate bending stresses. The notch-hinge on the height flexure consists of a 1 mm web between two circular notches of radius 4 mm. A finite element analysis indicates that when the manual adjuster is fully extended or retracted 150  $\mu\text{m}$ , the Von Mises stress in the hinge reaches 79 MPa, which is 29% of the yield stress of annealed type 403 stainless steel. This deflection causes an internal bending moment of 0.459 N·m in the notch hinge. A preload extension spring keeps the manual adjuster against the base of the flexure. An

analysis similar to that presented in Equation (1) indicates that a preload force of the extension spring of 33.3 N is more than adequate for this flexure.

A hardstop prevents the flexure from deflecting too far and plastically deforming the notch-hinge. To reach a Von Mises stress of 275 MPa in the notch hinge, the deflection of the hardstop must be 423  $\mu\text{m}$ . Therefore, hardstops limit the motion of the flexure to 423  $\mu\text{m}$ . Figure 10 illustrates a finite element analysis of the stress in the notch hinge when the flexure bottoms out on its hardstop. This deflection causes an internal bending moment of 1.60 N·m in the notch hinge. To prevent the notch hinge from plastically deforming, a safety factor is created by heat-treating the material at 980 °C and tempering at 650 °C to increase the yield strength to 586 MPa [2]. Therefore, the stress experienced when the height flexure bottoms out against the hardstop is 47% of the theoretical yield strength of the heat-treated stainless steel.

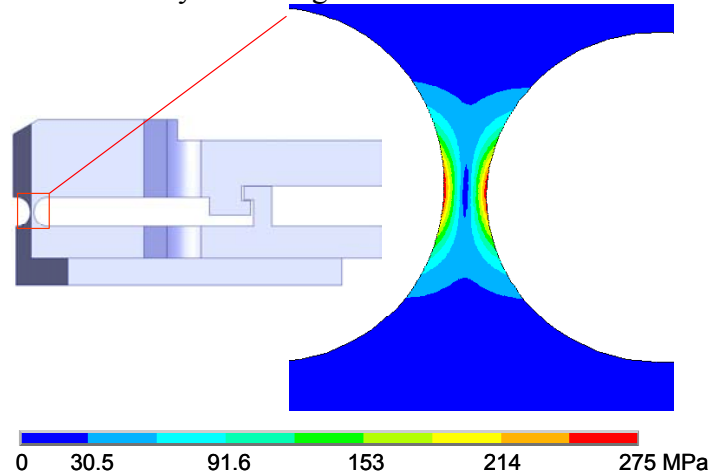
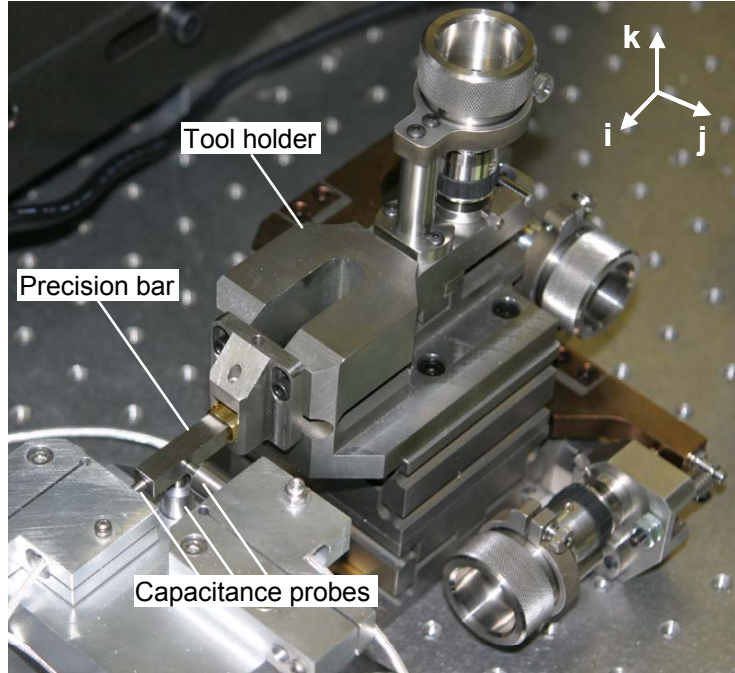


Figure 10. Finite element analysis of the Von Mises stress in the notch hinge of the height flexure

## PERFORMANCE OF THE TOOL HOLDER

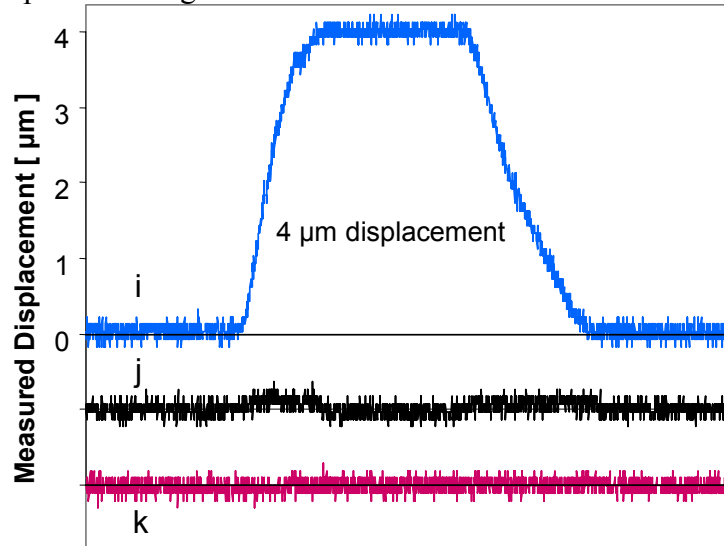
### Parasitic Errors

The performance of the tool holder was determined by measuring its motion using capacitance probes. A precision ground parallel bar was inserted into the tool holder in the location where a cutting tool would normally be mounted. Three capacitance probes measured the displacement of the bar in the i, j, and k directions, as shown in Figure 11. Each capacitance probe has a range of 50  $\mu\text{m}$ , a peak-to-valley resolution of 6 nm, and RMS resolution of 0.5 nm. The amplifier for each probe produces a voltage with a sensitivity of 0.4 V/ $\mu\text{m}$ , which was collected by a multi-channel digital storage oscilloscope.



**Figure 11. Measurement of tool holder performance using capacitance probes**

The cross-talk between the axes was determined by moving each actuator independently and measuring the displacements in the  $i$ ,  $j$ , and  $k$  directions. To make these measurements, the oscilloscope was set to a vertical scale of 1.0 V/major division, which corresponds to a resolution of  $0.1 \mu\text{m}$ . The manual micrometer adjuster connected to the  $i$ -motion flexure was rotated by hand to move the precision bar  $4 \mu\text{m}$  in the  $i$  direction. The data collected by the three capacitance probes is plotted in Figure 12.



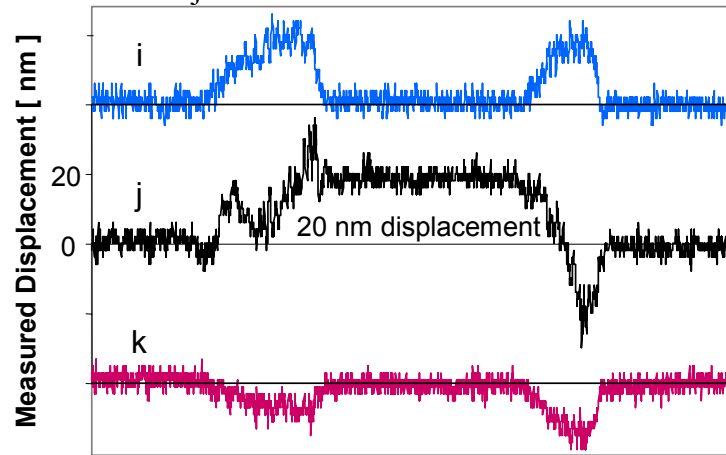
**Figure 12. Displacements measured by capacitance probes for a  $4 \mu\text{m}$  displacement in the  $i$  direction**

When the manual micrometer adjuster connected to the  $i$ -motion flexure was rotated, the motion in the  $j$  and  $k$  directions was less than the  $0.1 \mu\text{m}$  resolution of the system. As shown in Figure 12, small elastic displacements occurred as the manual micrometer adjuster was rotated, but no significant net displacements occurred in the  $j$  or  $k$  directions after the knob on the manual

micrometer adjuster was released. Similar tests revealed that the parasitic error motions that accompanied adjustments in the j and k directions were also less than the 0.1  $\mu\text{m}$  resolution of the system.

### Sensitivity

To determine the minimum increment of motion of the tool holder in the i, j, and k directions, each of the manual micrometer adjusters was rotated the smallest amount possible that produced a noticeable, controlled displacement. To make these measurements, the oscilloscope was set to a vertical scale of 20 mV/major division. Figure 13 contains a plot of the data measured by the three capacitance probes when the tool holder was moved approximately 20 nm in the j direction and then back to its original position. As the manual micrometer adjuster connected to the j-motion flexure was rotated, the tool holder deflected elastically in the i and k directions, but no noticeable net displacements occurred in the i or k directions after the knob on the manual micrometer adjuster was released.



**Figure 13. Displacements measured by capacitance probes for a minimal displacement in the j direction**

Similar tests were performed to determine the minimum increment of motion of the tool holder in the i and k directions. The minimum increment of motion of the tool holder in the i direction was also approximately 20 nm, and the minimum increment of motion in the k direction was approximately 13 nm.

### Stiffness

The flexure-based tool holder is not as stiff as some commercially available tool holders intended for four-axis diamond turning. One disadvantage of using flexures to achieve precise alignment of the cutting tool is that they have a relatively small stiffness [4]. To determine the stiffness of the flexure-based tool holder, a precision ground steel bar with a height of 9.5 mm and a width of 6.4 mm was used to mimic the shank of a cutting tool mounted in the tool holder. Forces were applied to the bar with a force transducer, and its deflection was measured with a capacitance probe. The forces were applied and the deflections were measured at a point on the bar located 15 mm from the end of the tool holder, which is the approximate location at which the cutting edge of a tool would normally be located. Therefore, the measured static stiffness represents an actual machining operation by including both the tool holder and the tool shank. The measured static stiffness of the system was approximately 8 MN/m in the i direction, 2



MN/m in the j direction, and 4 MN/m in the k direction. For many machining applications, the stiffness of the flexure-based tool holder is much larger than required.

There are some machining applications in which the stiffness of the tool holder is very important. For example, when attempting to diamond turn a workpiece with the best possible surface finish, the structural loop between the tool and the workpiece should be as stiff as possible. If the tool holder is the greatest source of compliance in the system, then its stiffness is particularly important. In applications for which the stiffness of the tool holder is more important than the tool setting accuracy, a commercially available tool holder may work well. One commercial tool holder that uses sliding plates preloaded with set screws rather than flexures was measured to be approximately 2 to 3 times as stiff as the flexure-based tool holder.

Often, the ability to set the tool with sub- $\mu\text{m}$  accuracy is far more important than maximizing stiffness. The stiffness of the tool holder is not an issue for many applications, such as when diamond turning with a small nose radius tool or with small depths of cut. The flexure-based tool holder has greatly facilitated the manufacture of meso-scale components with complex three-dimensional shapes with sub- $\mu\text{m}$  profile accuracies, such as components for laser targets.

## **SUMMARY**

A newly designed tool holder facilitates the machining of precision meso-scale components with complex three-dimensional shapes with sub- $\mu\text{m}$  accuracy on a four-axis lathe. Adjustments to the tool position are made using manual micrometer adjusters that drive flexure mechanisms, which were designed using finite element analyses. Two flexures are used to adjust the lateral position of the tool so that the center of the nose radius of the tool is aligned with the axis of rotation of the B-axis. Another flexure adjusts the height of the tool to align the cutting edge with the spindle axis of the machine tool. When making adjustments to the cutting tool position with this tool holder, the cross-talk between axes was less than the 0.1  $\mu\text{m}$  resolution of the measurement system. The tool holder has a minimum increment of motion of 20 nm. In addition, the tool holder does not contain any sliding surfaces, so there is no potential for trapped friction to be released and cause unwanted movement of the tool. Making an adjustment to the tool position requires the user to simply turn a manual micrometer adjuster, which is much simpler than the process required for some previously designed tool holders. This tool holder has improved the precision with which tools can be set, and it has significantly reduced the time required to set a tool.

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